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**ACCELERATED LIFE TESTS OF SPECIMEN HEAT PIPE FROM
COMMUNICATION TECHNOLOGY SATELLITE (CTS) PROJECT**

by Leonard K. Tower and Warner B. Kaufman
Lewis Research Center
Cleveland, Ohio 44135
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ACCELERATED LIFE TESTS OF SPECIMEN HEAT PIPE
FROM COMMUNICATION TECHNOLOGY
SATELLITE (CTS) PROJECT

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Lewis Research Center

SUMMARY

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A methanol-filled stainless-steel heat pipe of construction identical to one now flying on the Communication Technology Satellite (CTS) was life tested at accelerated conditions. After 14 200 hours of testing, equivalent to about 70 000 hours of flight conditions, the noncondensable gas inventory of the variable-conductance pipe increased by about 20 percent over the original charge. The theoretical increase based upon a fit of experimental data from an extensive study of methanol stainless steel pipes was 0.1 percent. The estimated increase of about 2.2 C in operating temperature resulting from the observed gas accumulation is insufficient to jeopardize the electronic gear which the heat pipes are intended to protect in the CTS.

Performance tests were conducted by determining for various heat inputs the elevation of the evaporator above the condenser required to cause evaporator dryout. The results agree well with theoretical predictions made by the manufacturer.

INTRODUCTION

Among the many uses advocated for heat pipes is spacecraft thermal control. One of the most recently flown spacecraft using heat pipes is the Communications Technology Satellite (CTS, ref. 1), launched January 19, 1976. The transmitter experiment package (TEP), a 200 watt travelling wave tube and associated power processor, was cooled by a variable-conductance-heat-pipe system designed by TRW. Heat dissipated by the

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TEP was transferred via three heat pipes to a radiator fin whence the heat was radiated to space. The requirements set by NASA Lewis Research Center were that the system should be capable of radiating a minimum of 196 watts from the radiator fin with one pipe failed when the evaporator saddle is 50 C. Whenever the saddle is less than 21 C the pipes should shut-off, with the conduction loss not to exceed 3 watts at a saddle temperature of less than 10 C.

To meet these requirements the contractor chose gas-loaded arterial heat pipes. The gas reservoir is wicked and is passive (no feedback control). Special priming caps are provided to vent noncondensable gas from the arteries. All parts of the pipes are stainless steel. The working fluid is methanol and the control gas is 90 percent nitrogen, 10 percent helium.

Heat pipes containing working fluids that are compounds may be subject to decomposition reactions between working fluid and container. This has been found true for stainless pipes containing methanol (ref. 2), nickel pipes containing water (ref. 3), and stainless pipes containing ammonia (ref. 4). The reactions result in the evolution of noncondensable gas, usually hydrogen, which is swept to the end of the condenser. Accumulating in sufficient quantity it partially blocks the heat rejection area of the condenser. For a given heat transport rate, the pipe is then forced to operate at a higher temperature which may be detrimental to the system being thermally controlled. A variable conductance heat pipe containing a gas reservoir and deliberately introduced gas, such as the CTS pipes, will be less sensitive to the decomposition evolution of noncondensable gas than a non-gas-loaded pipe.

The results of the research program (ref. 2) with methanol in stainless steel pipes indicated that the CTS heat pipes should experience no serious problem due to gas generation during the two year design life of the satellite. The time schedule for flight of the satellite did not allow a corroboration of this prior to design of the system. However, during the period of assembly of the satellite, life tests of specimen CTS pipes were initiated at the Lewis Research Center of NASA. One of these tests, at an accelerated condition, is described in this report.

APPARATUS AND PROCEDURE

The pipe chosen for the accelerated life test was the shortest of the three in the CTS array. A sketch of the heat pipe is shown in figure 1. The total distance along the centerline of the pipe from the gas reservoir to the end of the evaporator is 172.7 cm (68 in.). The manufacturer, TRW, used the same materials, processing, and construction methods as were employed in manufacturing the CTS variable conductance heat pipe system. All parts were of 304 stainless steel except for the screen used for the arteries. The arteries were constructed of 150 mesh (per in.) 316 stainless. The metal felt slab wick had a porosity of about 0.84. The pipe was spirally threaded with about 39 grooves per cm.

The pipe was delivered with 70.72 grams of spectrophotometric grade methanol and was nominally charged with 1.79×10^{-3} g moles of a mixture containing 90 percent nitrogen and 10 percent helium, both of research grade.

In the tests reported herein the pipe was mounted on a support plate (fig. 2) pivoted about an axis lying in the plane of the heat pipe, intersecting the center of curvature of the 90° bend, and making a 45° angle with each leg (fig. 1). Rotation about this axis raised or lowered the evaporator relative to the condenser. The pipe was enclosed in a vacuum housing which was evacuated to about 6N/m^2 (45 millitorr).

The evaporator heater assembly is shown in figure 3. Initially, a radiation heater (fig. 3(a)) was used to heat the evaporator. The heater was a stainless steel tube of 2.5 cm (1 in.) diameter, 31.8 cm (12.5 in.) long, sprayed with alumina and wound with resistance wire. The outside of the heater was shielded with several wraps of stainless steel foil to cut down radiation losses. The condenser rejected heat to two heat extractors of 32 brass blocks each, segmented to facilitate manufacture and assembly on the heat pipe. The blocks, 2.5 cm (1 in.) long, were penetrated by two stainless steel tubes of 0.63 cm (0.25 in.) outside diameter (fig. 4). The blocks were spaced 0.25 cm (0.1 in.) apart on the tubes and were clamped together on the condenser by bolts through each block. The heat extractors were spaced off the condenser by 0.030 cm (0.012 in.) thick shrink plastic tubing to facilitate alignment and provide a thermal resistance. The heat

extractor covered the entire straight portion of the condenser. In the CTS system the radiating panel includes a portion of the pipe between the bend and evaporator, so the condenser is somewhat longer and the adiabatic section somewhat shorter than in the comparable pipe described here.

The tubes through the heat extractors were manifolded together in series so that coolant circulation through them resulted in a uniform average sink temperature over the length of the condenser. The cooling system circulated water with temperature controlled within about $\pm 1\frac{1}{2}$ C at a rate of about 1 kg/min (2.2 lb/min).

A small resistance heater and a water-cooled sink were strapped to the gas reservoir (fig. 4). These enabled the gas reservoir temperature to be varied independently of the condenser.

For some of the performance checks and gas accumulation tests, a temperature-controlled bath designed for hospital and laboratory use, containing ethylene-glycol water mixture, was used as a source of coolant. Temperatures below -10 C, controlled to ± 0.1 C, were obtained with this system.

Temperatures were measured initially by means of 28 iron constantan thermocouples spot welded to the pipe at the positions shown in figure 5. The wires to the couples in the evaporator had glass insulation because of the high radiation temperature of the heater. These wires were strung along the pipe to the end of the heater. They were held in place by narrow bands of aluminized tape, some of which were also placed to cover the junction beads.

The couples in the condenser were located under the shrink tubing. Initially there were ten couples on the evaporator, four on the adiabatic section, and thirteen on the condenser. Three of the evaporator couples were at the bottom and seven at the top of the pipe. Couples in the adiabatic section and condenser were all on the top of the pipe.

After about 7500 test hours had elapsed the radiant heater was replaced. This heater was employed originally to permit easy attachment of the thermocouples to the pipe. Also, it was felt that the absence of a massive heater block would result in a more rapid response to dry out conditions.

Unfortunately this proved to be precisely the case. When dry out occurred the over-temperature relay was activated, but the mass of the radiant heater enabled it to continue radiating to the dried-out region of low-heat capacity for some time. These over-temperature conditions became a source of some concern. Subsequent analysis showed that the excursions had little effect on the volume of gas generated during the life test. Nonetheless the problem of excursions provided a motive for eliminating the radiant heater. Further motive was furnished by a measurement of the heater power loss to the vacuum jacket by radiation. A simple analysis of the thermal transients in the condenser blocks occurring at start-up of the pipe was conducted. With heater input power of 130 watts, the radiation loss was computed to be nearly 50 percent. Among other things this made calculation of the heat pipe thermal transport less accurate. Obviously a better method was required for thermally coupling heater and pipe.

The heater was converted to the conduction mode by the insertion of a 30.5 cm (12 in.) long split annular aluminum sleeve (fig. 3(b)). This sleeve was inserted between the heater and pipe. Before assembly all surfaces of contact between the sleeve and the heater or pipe were thoroughly coated with conduction grease to reduce thermal resistance.

The sleeve was fabricated in two longitudinal segments, with a thin insulating spacer disk between. The shortest segment, 5 cm or 2 in., was toward the evaporator end. The segmenting was intended to provide partial thermal insulation of the beginning of the evaporator, so that the temperature rise associated with dry out could be detected more readily.

Use of the sleeve required new evaporator thermocoupling. Wires were laid along slots in the outside of the annulus segments (fig. 3(b)). The junctions were flush with the inner surface of the sleeve. The short segment contained one couple and the long segment two couples positioned on top of the pipe and located as indicated in figure 5.

A calorimetric study was conducted to determine the power rejected to the cooling water relative to the power into the heater, when the conduction sleeve was employed. Because of the very small temperature rise in the cooling water, an accurate heat balance could not be made. However, the

calculated radiation loss from the heater, based on measurements of the heater surface temperature, was less than three watts. Therefore, the power input to the heater was used as the power transported by the heat pipe.

Data were acquired by means of a calculator-controlled data system designed to handle a number of simultaneous heat-pipe tests. This system consists of a digital voltmeter and scanner coupled by appropriate input and control cables to a programmable calculator. A tape cassette memory device is also coupled to the calculator. The thermocouple lines terminate in two connectors, into which lines from the scanner can be plugged.

A program was devised for the calculator to perform the following functions by command from the keyboard: verify that the data tape in the cassette memory belongs to the heat pipe under test; give visual display of one thermocouple temperature, or all of them, one at a time; take data and process for recording on tape; print out data; display input power to heater; record data on tape.

Automated plotting of the taped data is accomplished with a desk top computer-controlled plotter. A representative plot is shown as figure 6. The entire data-processing assembly can be wheeled about to serve many heat pipes.

RESULTS

Performance tests. - When about 11 300 hours had elapsed, the heat transport capacity of the pipe was determined for three pipe temperatures (defined as the average of the four adiabatic section thermocouple readings). The temperature-controlled bath was used as the source of coolant for the condenser and gas reservoir. A small amount of heat was supplied by the gas reservoir heater to assure that the reservoir and the inactive portion of the condenser were within $\pm 1/2$ C of each other. The data taken could then be compared to theoretical performance computed for the pipe.

Figure 7 shows maximum heater power against evaporator elevation for the pipe. Figures 7(a), (b), and (c) pertain to adiabatic section temperatures of 40 C, 50 C, and 60 C, respectively. Solid circles indicate

that dryout was experienced, as evidenced by runaway increase of any of the evaporator thermocouples. Open circles represent, for any power level, the maximum tilt at which the evaporator temperatures remained stable. The data were uncorrected for any power radiated from the heating element to the vacuum housing (in this case conduction rather than radiation was the mode of transport from heater to pipe, so power loss was small).

The data package furnished with the heat pipe by the manufacturer presented a theoretical curve of heat pipe capacity against temperature for an evaporator elevation of 0.64 cm (0.25 in.). These data at temperatures of 40, 50, and 60 C are shown in figures 7(a), (b), and (c) as solid squares. They fall close to extrapolations of linear least mean square fits of the experimental data.

Gas accumulation. - During virtually all of the time the pipe was in operation the circulating cooling system was set to maintain the pipe at about 60 C, determined at the adiabatic section. Relative to temperatures in the duty cycle experienced in the satellite, this represents a condition for accelerated formation of noncondensable gas. Only during the brief periods when performance checks were being made was the pipe operated at any temperature except 60 C.

The pipe was charged initially with 1.79×10^{-3} g moles (3.94×10^{-6} lb moles) of noncondensable gas. At the time of start-up of the pipe, no tests were undertaken to verify the amount of gas present. However, after 14 200 hours of accelerated life testing, a series of tests were conducted which give evidence as to the noncondensable gas inventory at that time.

These tests were divided by operating procedure into two groups (table 1). In group A, the gas reservoir temperature was maintained as near as possible to the temperature of the inactive portion of the condenser (that portion on the cold side of the gas front). The coolant bath temperature was maintained constant in a sequence of tests at approximately 1, 10, 20, 30, 40, and 50 C. At each bath temperature level, temperature profile data were taken at approximately 11, 21, and 31 watts. At these conditions, the gas front was located between thermocouples 12 and 13, figure 5, as determined by the drop in temperature level between the couples. These couples are

spaced relatively close, minimizing error due to uncertainty in front location. The end of the evaporator was 0.33 cm (0.125 in.) below the end of the condenser for all runs in group A.

The data in group B were taken with the gas reservoir heated to approximately the adiabatic section temperature. In this case the gas reservoir acted also like an evaporator. Most of the noncondensable gas must then reside in the condenser.

For group B, the evaporator power was set at about 15 watts, and the coolant bath temperature was 30 C. The end of the evaporator was 0.33 cm (0.125 in.) below the end of the condenser for the first three gas checks of group B. For the last three runs the evaporator end was 0.95 cm (0.375 in.) above the condenser end. The change in elevation was intended to alter the location of any excess fluid from evaporator to condenser.

The noncondensable gas inventory in the heat pipe can be represented

$$n = \sum_{i=1}^{32} \frac{p_{va} - p_{vi}}{RT_i} L(A - A_{li}) + \frac{p_{va} - p_{vr}}{RT_r} (V - V_{lr}) \quad (1)$$

where

p pressure

T wall temperature determining the local vapor pressure

L length of condenser beneath a heat extractor block, plus gap width between two blocks

A internal cross section area of heat pipe excluding wick and artery

A_l area of liquid

V volume of gas reservoir exclusive of wick, artery, and screen

V_{lr} volume of liquid in reservoir

R universal gas constant

Subscript s:

- a adiabatic section
- i number of a block
- r reservoir
- v vapor

Here A was computed from TRW prints as 0.99 cm^2 (0.154 in.^2). Likewise V was 148 cm^3 (9.04 in.^3). Also L was 2.74 cm (0.18 in.). The source of an expression for methanol vapor pressure as a function of temperature was reference 6.

One big uncertainty in applying equation (1) is the distribution of fluid along the wick. The assumption was made that the area occupied by the fluid was uniform for the length of the pipe. Moreover the assumptions were also made that the distribution of fluid between heat pipe and gas reservoir remained constant and that fluid density variations could be ignored for the data of table I. In this case the liquid area A_l at any section, and the liquid in the reservoir can be related to the total charge volume V_l :

$$V_l = 172.7 A_l + V_{lr} \quad (2)$$

where 172.7 cm (68 in.) is the length of the pipe exclusive of the reservoir. The mass of methanol in the vapor phase is negligible relative to the charge. The fluid charge was 70.72 gms . At a density for methanol of about 0.78 gms/cc over the range of temperatures indicated for the pipe in table I, V_l is 90.67 cc . Substitution of equation (2) in equation (1) gives

$$n + (K - 172.7 K_r) A_l = K A + K_r (V - V_l) \quad (3)$$

where

$$K = \sum_{i=1}^m \frac{p_{va} - p_{vi}}{RT_i} L$$

$$K_r = \frac{p_{va} - p_{vr}}{RT_r}$$

This equation is of the standard form for a straight line, $y = ax + b$ if the following identifications are made:

$$n = b$$

$$a = A_l$$

$$x = K - 172.7K_r$$

$$y = KA + K_r(V - V_l)$$

Table I contains a tabulation of K , K_r , x , and y computed for the data of groups A and B. Small negative values of K_r are taken as zero since p_{vr} cannot materially exceed p_{va} . If T_{vr} exceeds T_{va} by any amount exceeding random instrument noise, dryout of the gas reservoir would be indicated, and the data would have to be discarded.

The assumption was made that the thermocouples on the exterior wall of the pipe gave the true vapor temperature of the interior. This assumption is best in the inactive portion of the heat pipe where thermal gradients through the liquid film and pipe wall due to heat transfer are least.

There were many block locations in the condenser with no thermocouples. For the data of group A, an estimated temperature for the inactive portion of the condenser was provided from a simple average of all the couples in that region. The temperatures of the blocks between couples 12 and 13 were estimated by assuming a linear decline between these points. This decline was extrapolated in blocks past couple 13 until the mean temperature of the inactive condenser was reached. When couples 12 and 13 were close in temperature indicating that much of the front was beyond couple 12, the data were not used.

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For data of group B an estimated temperature was provided for each block from values at the two nearest couples on either side. The increase or decrease was adjusted to be linear with distance between the couples. The data of group B were adjusted in this manner since separate heating of the gas reservoir resulted in elevated temperatures at both ends of the condenser. The inactive portion of the condenser was therefore more difficult to define than for group A.

These methods of handling the front are admittedly imprecise since the front is known from analytical and experimental work to have a non-linear form (ref. 6). However, the length of the front in this heat pipe was observed to be only about four inches. The assumption of linearity probably causes no more error than the uncertainty of knowing the location of the front from temperatures at couples 12 and 13.

A least squares fit of the data in table I yields $A_2 = 0.425 \text{ cm}^2$ or 0.0659 in.^2 and $n = 2.15 \times 10^{-3} \text{ g moles}$ or $4.74 \times 10^{-6} \text{ lb moles}$. A plot of x against y showing both the data from table I and the least mean squares fit is shown as figure 8. Evaporator elevation change from -0.32 to $+0.95 \text{ cm}$ is seen to have no significant effect.

The initial charge of noncondensable gas at TRW was $1.79 \times 10^{-3} \text{ g moles}$ or $3.94 \times 10^{-6} \text{ lb moles}$. The apparent increase in gas at the 14 200 hour point of testing was thus $3.6 \times 10^{-4} \text{ g moles}$, or 20 percent.

DISCUSSION

Comparison of observed and theoretical gas generation. - Reference 2 contains an analysis of gas generation in methanol, stainless-steel heat pipes derived from experimental data. Expressions contained therein enable a theoretical estimate of noncondensable gas to be made for the heat pipe discussed herein.

The investigators of reference 2 found that their data could be fitted by two expressions. One with parabolic time dependence governed the initial stages of corrosion:

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$$\Delta n = B_1 A t^{1/2} e^{-Q_1/kT} \quad (4)$$

while the later stage of corrosion had a linear dependence:

$$\Delta n = B_2 A t e^{-Q_2/kT} \quad (5)$$

where

Δn gas formed by corrosion, g moles

A total internal area of stainless steel contacting methanol, cm^2

t time, hrs

B_1 2.57×10^{-6} g moles/hr $^{1/2}$ cm^2

B_2 2.66×10^{-7} g moles/hr cm^2

Q_1/k 4368 C

Q_2/k 19 339 C

The time t_c at which the rates become equal for the two regimes is given by

$$t_c = \frac{B_1^2}{4B_2^2} \exp^{2(Q_2 - Q_1)/kT} \quad (6)$$

At 60 C this time is 2.5×10^{12} hours, so the entire period of the present test was in the regime of quadratic time dependency.

The internal area of the heat pipe herein was estimated to be 3077 cm^2 (477 in.^2) from material specifications furnished by the manufacturer. At 60 C and 14 200 hours the evolved gas is computed from equation (4) to be 1.9×10^{-6} g moles, an increase of 0.1 percent over the original inventory. The observed increase was 3.6×10^{-4} g moles, a 20 percent increase.

This observed increase is likely to be on the high side of the "theoretical" increase in gas for two reasons. One reason is that wall temperatures rather than vapor temperatures are used in equation (1). Because of heat transfer, the wall temperature will be below the interior temperature determining the vapor temperature in the active portion of the condenser.

The other reason for expecting observed gas to be high is temperature excursions. These resulted from the use of the radiant heater. During the period of some 7500 hours when the radiant heater was used, dryout during performance tests resulted in temperature excursions. The radiant heater, operating at an estimated temperature of less than 700 C, had sufficient heat capacity to continue radiating to the evaporator although the over-temperature limit switch was set to cut heater power when a 10 C increase in evaporator temperature occurred.

About thirty such excursions were encountered. The general practice when the operator observed dryout was to raise the condenser end of the heat pipe and flood the evaporator with liquid. Seldom did the thermocouple temperatures displayed on a bar-graph oscilloscope exceed 100 C during this procedure. The length of time for the temperatures to fall to normal after flooding with fluid was about 10 seconds.

An order of magnitude can be placed on the effect of such excursions by the use of equation (4). The amount of gas evolved in an increment of time δt at time t is obtained by differentiating equation (4) with respect to time:

$$\delta(\Delta n) = \frac{B_1 A}{2} t^{-1/2} e^{-Q/kT} \delta t \quad (7)$$

The amount of gas formed per unit time decreases as t increases. This effect is attributed to the penetration of a passivating film during the initial phase of corrosion (ref. 2).

Assume a case far worse than that encountered in actuality, an excursion to 400 C at a test time on the pipe of one hour. Also assume that all 30 excursions actually experienced occurred in succession at this time,

and had a duration of 10 seconds each, for a total of 5 minutes at 400 C. By equation (7), 5×10^{-7} g moles or 1.1×10^{-9} lb moles of gas would have been formed by this extreme excursion. This is but 0.1 percent of the amount computed to have been formed.

The temperature excursions thus appear to have had little effect upon the experimental observation of gas accumulation in the life test.

Time scaling factor between accelerated life tests and pipe in-flight service. - The 14 200 hours at the accelerated life test condition can be converted to the time required for the same gas accumulation in the CTS pipes, assuming the validity of equation (4) for temperature-time dependency only.

During the period from February 8, 1976 to May 5, 1977 the heat pipe on the satellite equivalent to the pipe described herein operated about 5612 hours at an average adiabatic section temperature of 33 C. The remainder of the time, about 5236 hours, a saddle heater maintained a temperature of about 20 C.

The duty cycle was actually irregular, depending on the needs of the users of the communications gear. For simplicity, assume that it was periodic over the entire time span, with a period of one day. Thus 12.42 hours were at an average of 33 C, and 11.58 hours were at an average of 20 C. In this case the differential form, equation (7), can be integrated to give

$$\Delta n = B_1 A \left\{ \sum_{m=0}^N \left[\sqrt{24m + \Delta t} - \sqrt{24m} \right] e^{-Q/kT_a} + \left[\sqrt{24(m+1)} - \sqrt{24m + \Delta t} \right] e^{-Q/kT_b} \right\} \quad (8)$$

where

N total number of days

Δt length of time at initial daily temperature, days

Equating (8) to equation (4) gives

$$t_t^{1/2} = \sum_{m=0}^N \left[\sqrt{24m + \Delta t} - \sqrt{24m} \right] e^{-Q/kT_a} + \left[\sqrt{24(m+1)} - \sqrt{24m + \Delta t} \right] e^{-Q/kT_b} \quad (9)$$

where t_t is time at life test temperature. Iteration of equation (9) for t_t of 14 200 hours gives N of 2903 days or 69 700 hours. The life test was thus equivalent to about 8 years of flight.

Change in operating temperature due to gas accumulation. - The increase in noncondensable gas due to corrosion will cause the heat pipe at a given power level to run hotter. The change in temperature for the heat pipe in this study can be estimated for a typical operating condition. The flat front theory of reference 5 can be used if a sharp interface between vapor and noncondensable gas is assumed. The heat transfer rate Q can be computed for a variable conductance heat pipe with reservoir and sink at the same temperature as (ref. 5)

$$Q = hA'(T_{va} - T_s) \left[L_c + \frac{V - V_{lr}}{A_v} - \frac{nRT_s}{A_v(p_{va} - p_{vs})} \right] \quad (10)$$

Here

L_c condenser length

A_v vapor cross section area of condenser

T_s sink temperature

p_{vs} vapor pressure at T_s

h overall heat transfer coefficient

A' heat rejection area per unit of condenser length

T_{va} adiabatic section temperature

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Assuming the liquid distribution found previously from the gas generation analysis, A_v is 0.567 cm^2 or 0.0879 in.^2 and $(V - V_{lr})$ is 118.6 cm^3 or 7.982 in.^3 .

When the heat pipe is at full power the last two terms in brackets are equal, or

$$V - V_{lr} = \frac{nRT_s}{(p_{va} - p_{vs})} \quad (10)$$

For an evaporator temperature of 50 C or 112 F , the requisite sink temperature is 28.88 C for the original charge of noncondensable gas, $1.79 \times 10^{-3} \text{ g moles}$.

If the gas inventory is now adjusted to the value after 14 200 hours of accelerated life testing, $2.15 \times 10^{-3} \text{ g moles}$, with Q and T_s held the same, equation (10) gives a new adiabatic temperature of 52.23 C . The rise due to evolved noncondensable gas under the specified conditions would thus be 2.2 C (4.0 F) in 14 200 hours of accelerated testing, equivalent to 69 700 hours of flight.

CONCLUDING REMARKS

Life and performance tests have been conducted on a heat pipe identical to one now on the Communications Technology Satellite. These tests verify that the noncondensable gas formed in the flight pipes is insufficient to endanger the electronic gear being cooled. The heat pipe performed to the predictions made by TRW. On the basis of this accelerated life test, the useful life of the CTS heat pipes will exceed considerably the two year program for the satellite.

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TABLE I. - GAS ACCUMULATION DATA

Group	Approximate bath temperature, C	Approximate adiabatic section temperature, C	Evaporator elevation, cm	Power, w	K, g mol/cm ²	K _r , g mol/cm ³	x, g mol/cm ²	y, g mol
A	1	38.4	-0.125	11	1.022×10 ⁻³	1.183×10 ⁻⁵	-1.021×10 ⁻³	1.693×10 ⁻³
	1	38.3	↓	20	.989	1.195	-1.075	1.667
	1	38.4	↓	32	.983	1.193	-1.077	1.660
	10	40.1	↓	12	.982	1.157	-1.016	1.638
	10	40.9	↓	20	.998	1.200	-1.075	1.679
	20	44.0	↓	12	.986	1.182	-1.055	1.657
	20	44.4	↓	21	.986	1.207	-1.098	1.672
	30	48.3	↓	11	1.007	1.240	-1.134	1.711
	30	48.5	↓	21	.998	1.264	-1.186	1.716
	40	53.2	↓	12	.977	1.257	-1.194	1.691
	40	54.5	↓	21	.979	1.277	-1.227	1.705
	50	59.7	↓	11	.998	1.210	-1.092	1.684
B	30	69.1	-0.125	16	3.316	.216	2.944	3.414
	30	72.0	↓	↓	3.738	.073	3.611	3.752
	30	70.3	↓	↓	3.549	(a)	3.549	3.522
	30	73.0	+0.375	15	3.918	(a)	3.918	3.888
	30	70.2	↓	↓	3.818	(a)	3.818	3.789
	30	68.5	↓	↓	3.747	(a)	3.747	3.718

^aSmall negative value K_r is set equal to zero in computing x and y.

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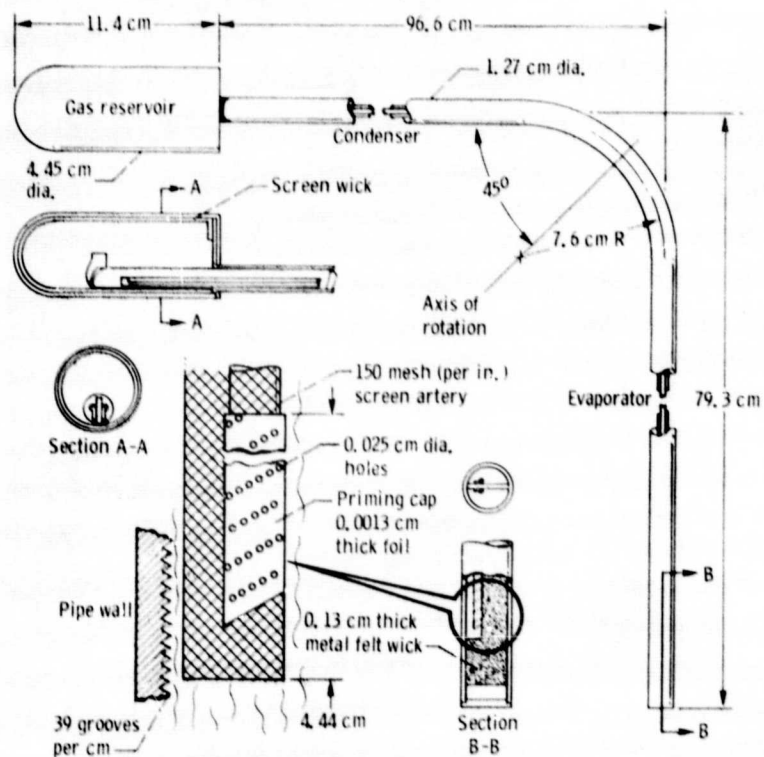


Figure 1. - Configuration of the CTS heat pipe with details of the pipe cross section, gas reservoir, and artery priming cap.

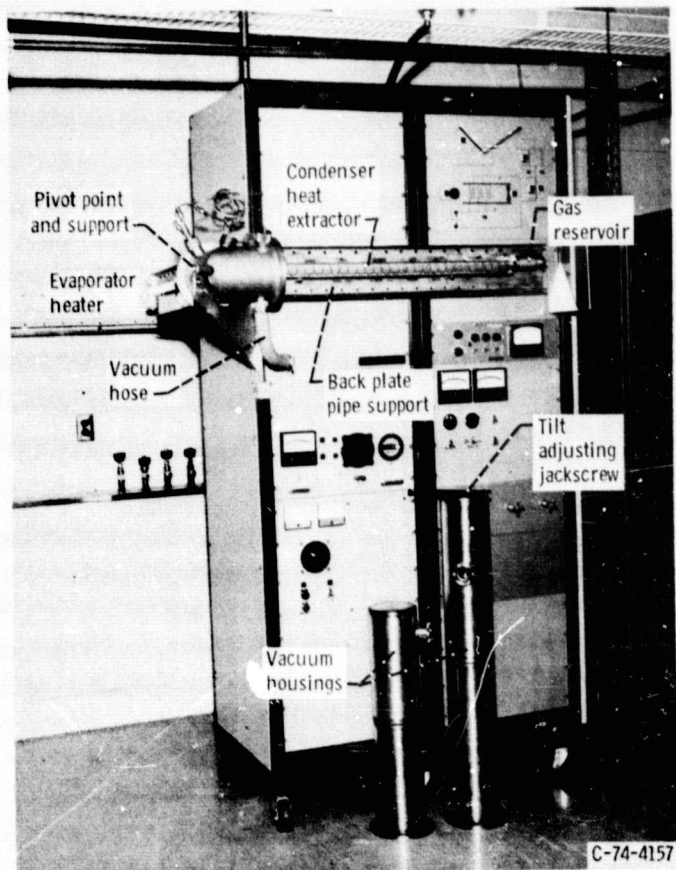
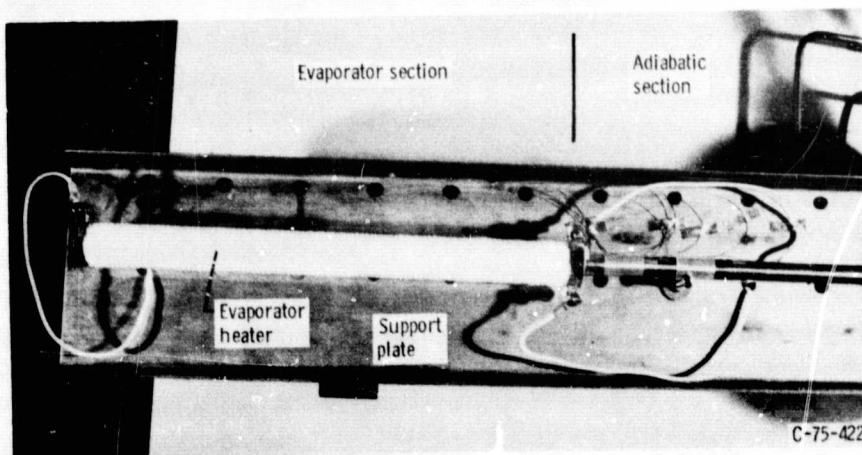


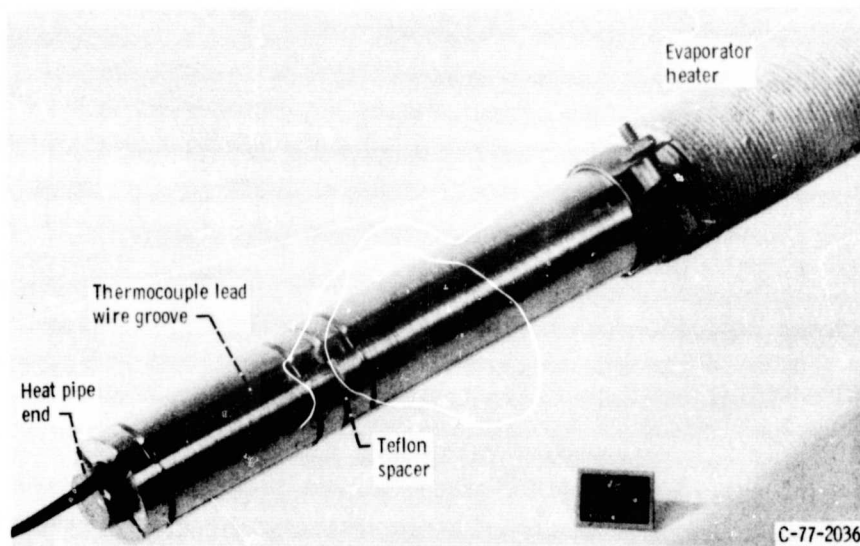
Figure 2. - Heat pipe test stand and equipment.



(a) Radiation heater.

Figure 3. - Evaporator heater assembly.

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(b) Annular conduction sleeve withdrawn from heater.

Figure 3. - Concluded.

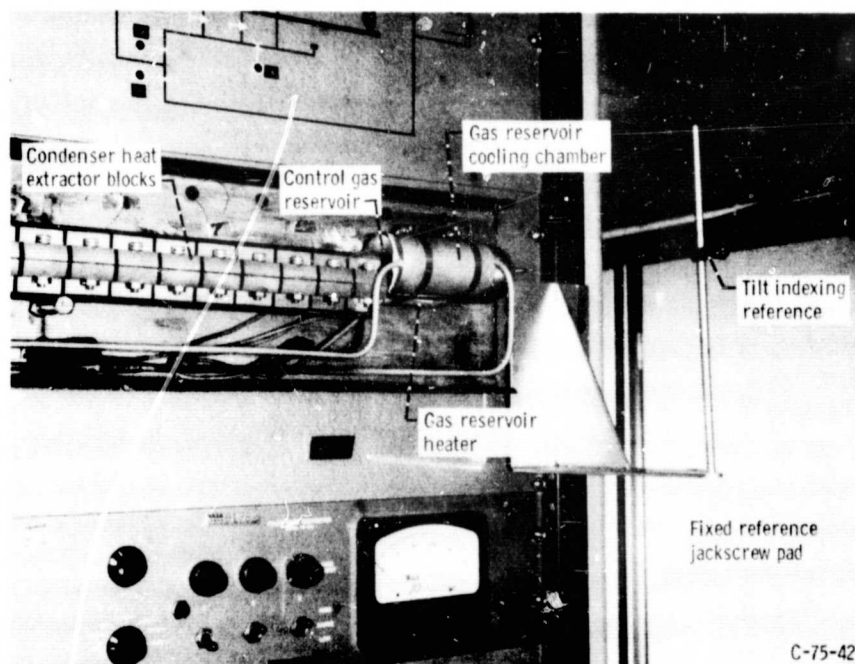


Figure 4. - Condenser heat extractor.

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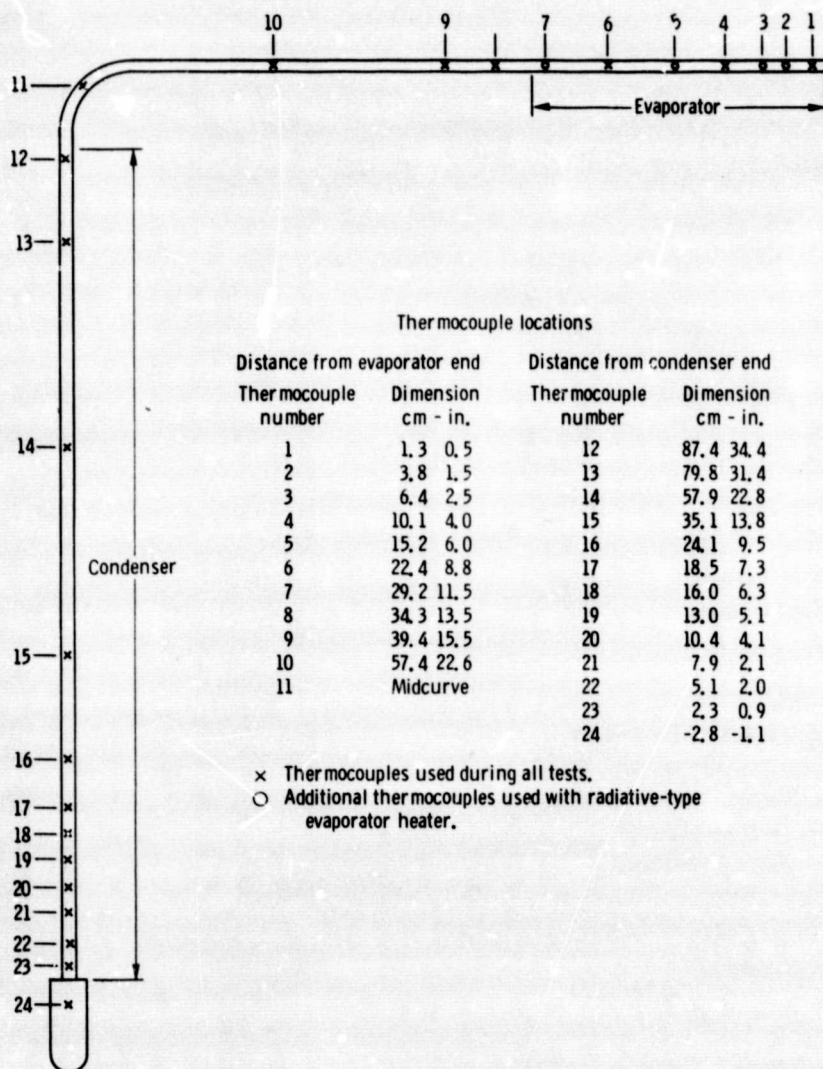


Figure 5. - Thermocouple locations.

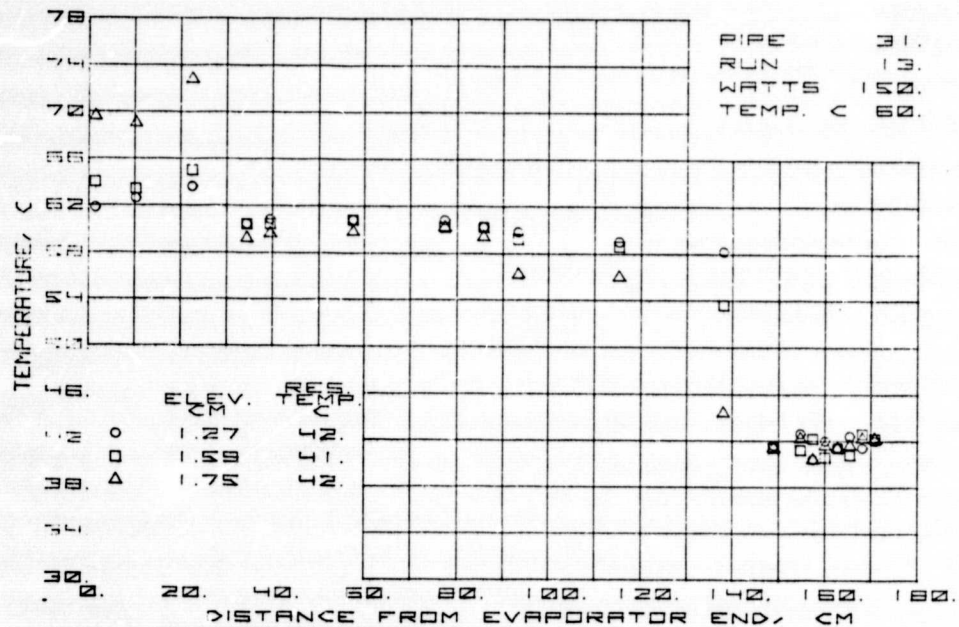


Figure 6. - Representative data plot produced from tape cassette by calculator-controlled plotter.

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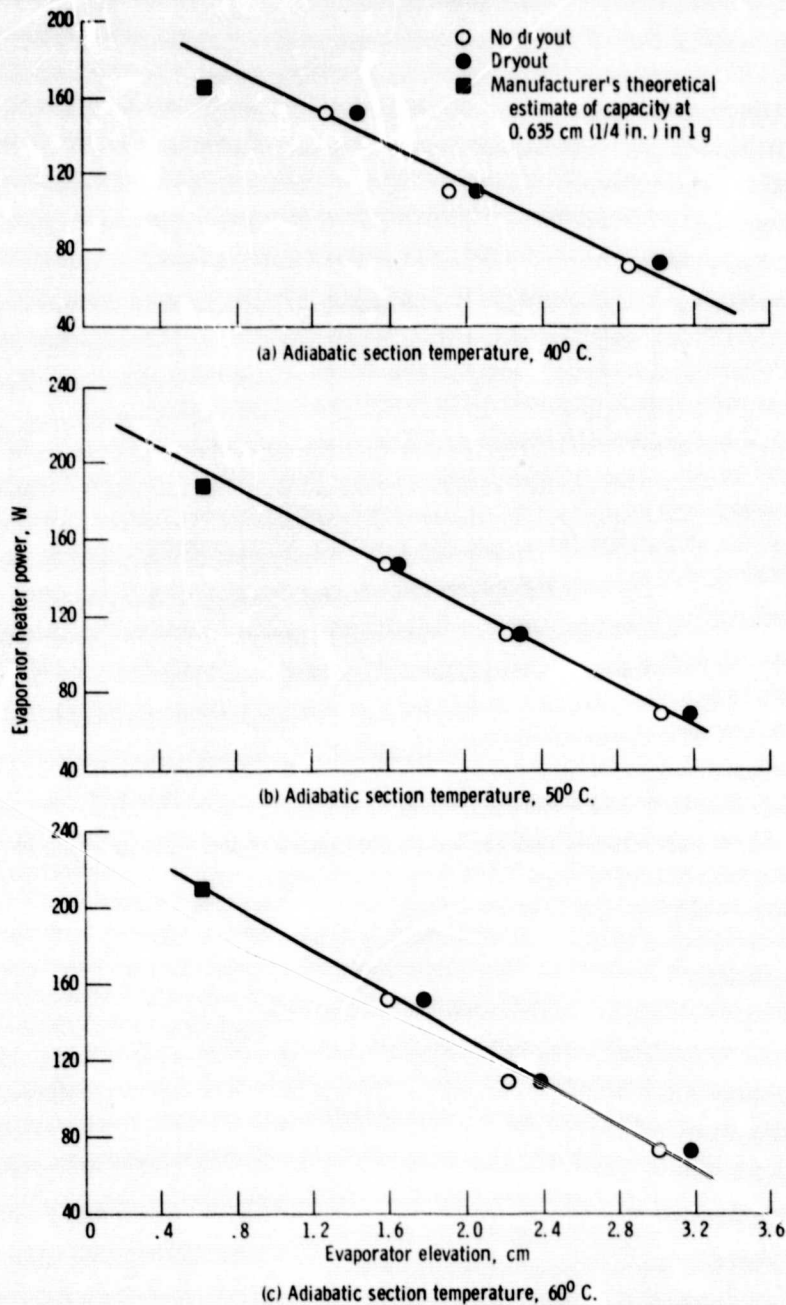


Figure 7. - Maximum evaporator heater power without dryout, as a function of evaporator elevation.

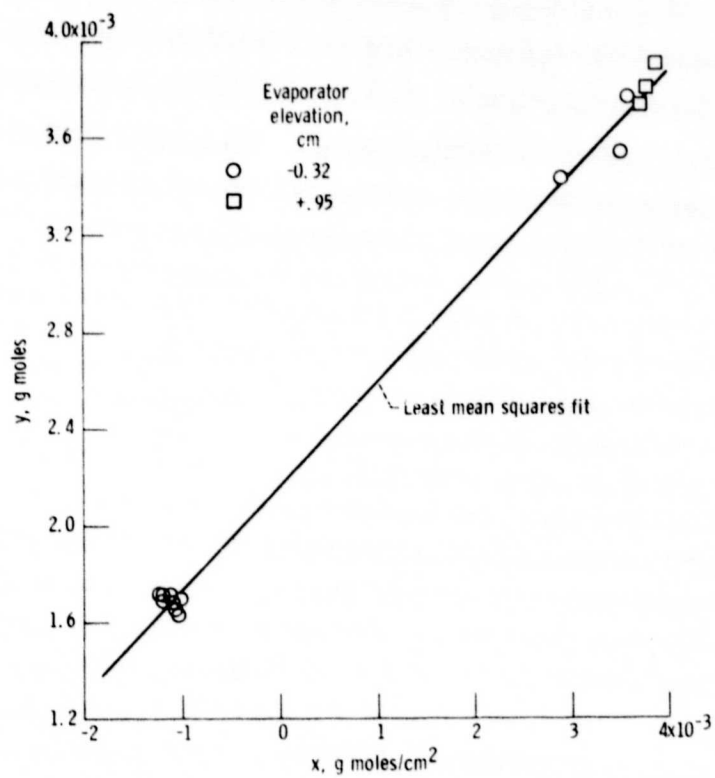


Figure 8. - Parameter x against y for data of table I, with least mean squares fit.